

Heat Transfer in the Condensation of Vapors

Including the Condensation of Diphenyl
and the Concentration of
Caustic Soda

BY

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The Condensation of Vapors^{1,2}

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Several important theoretical contributions have been made to the subject of vapor condensation that have not been generally recognized. The contributions of Nusselt in particular show the quantitative effects of physical properties of the vapor and liquid, impurity, superheat and velocity of the vapor, and the size and shape of the surface on the transmission of heat from condensing vapors. This theory is briefly reviewed.

Application of the Nusselt theory to data on vapor condensation on the outside of horizontal tubes shows that the theory is valid for this case.

Application of the theory to data on vertical tubes shows that on long tubes, or at high temperature differences, the theory does not hold, probably owing to turbulence and drop formation. An equation for determining the start of turbulence in a film of condensate is derived, and it is shown that turbulence accounts for most of the observed deviations from the Nusselt theory.

The effects of superheat, vapor velocity, and impurities are pointed out. Several corrections to the Nusselt theory are made, especially on the effect of vapor velocity.

A CRITICAL survey of the literature on heat transfer from condensing vapors to solid surfaces shows that this subject is extremely complicated theoretically, and

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that experimental results to date seem to be rather inconsistent and incomplete.

A large amount of work has been done on the transfer of heat from steam to water through a metal wall, but very few investigators have measured the temperature of the metal. Therefore, in most cases only over-all coefficients of heat transfer have been obtained, which could be changed by varying conditions on either side of the metal boundary. A rough calculation of the metal surface temperature could be made if the coefficient of heat transfer for the water side were known, but such calculations are never accurate and may lead to erroneous conclusions.

The subject of heat transfer from condensing vapors to solid surfaces has not received so much attention as that from a solid surface to a non-boiling liquid because of the experimental and theoretical difficulties. Moreover, until very recently it was not necessary to know very much about this subject, because the resistance to heat flow of the condensing vapor has been much less than the other resistances involved and often could be neglected. The application of dimensional analysis to the problem is extremely difficult because of the large number of variables and because there is a change of state from vapor to liquid.

Very recently, however, liquor velocities in evaporators and condensers have been increased to such a point that the vapor resistance is becoming of greater importance. In fact, in the experiments by Badger, Monrad, and Diamond (4) on the evaporation of caustic soda with diphenyl vapor, the major resistance to heat flow was on the vapor side, and the liquor resistance could be almost neglected in calculating the heat flowing. This means that any appreciable increase in the amount of heat transferred must be made by changing conditions on the vapor side.

Nomenclature

The writers have thought it advisable throughout this paper to use the symbols found most generally in American practice and at the same time not to use the same symbol for more than one property. In general the properties of the liquid are used without subscript and those of the vapor symbolized with the subscript v .

For the most part all physicists and other scientists use the c. g. s. system for the units involved, but no such agreement is found in technical practice. The European or technical metric system was for a long time based on the meter, kilogram, large calorie, second, etc., and conformed in general to the c. g. s. system except that the units of force were defined in kilograms, corresponding to grams instead of dynes in the c. g. s. system. This had a marked effect on the units of various other properties such as viscosity. More recently a mixed set of units has been used in which the hour replaced the second for many of the properties. This has led to considerable confusion.

The English system of units is in even worse condition. Very few workers use the same units, and in general all use a mixed system. Thus a very familiar example is the common Reynolds' criterion, which is supposedly a dimensionless group.

In the English system this is usually

$$\frac{Dup}{z} = \text{inch} \times \frac{\text{foot}}{\text{second}} \times \frac{\text{pound}}{\text{cubic foot}} \times \frac{\text{second-centimeters}}{\text{gram}}$$

SYMBOL	VALUE	UNITS (TECHNICAL METRIC)
α	$\frac{L\rho^2}{4kz(t_v - t_w)}$	
A	$\frac{L\rho^2 k^2}{z}$	
b	$\frac{C_3 w^2 \rho v}{3\rho}$	
B	Width	m.
C	Constant	
D	Diameter of tube	m.
E	$\frac{C_3 \rho v}{3\rho k}$	
f	Friction factor	
g	Acceleration of gravity	$\frac{\text{m.}}{\text{sec}^2} = 9.81$
G	Weight of condensate	kg.
H	Height of wall	m.
k	Thermal conductivity	$\frac{\text{Cal.}}{\text{m.}^\circ \text{C. sec.}}$
l	Tube length	m.
L	Latent heat	$\frac{\text{Cal.}}{\text{kg.}}$
m	Hydraulic radius	m.
P	Pressure	$\frac{\text{kg.}}{\text{m.}^2}$
Q	Heat flowing	$\frac{\text{Cal.}}{\text{m.}^2 \text{ hr.}}$
r	Radius	m.
S, S_1, S_2	Friction force	$\frac{\text{kg.}}{\text{m.}^2}$
t, T	Temperature	$^\circ \text{C.}, ^\circ \text{A.}$
u	Velocity, x direction	$\frac{\text{m.}}{\text{sec.}}$
U	Coefficient of heat transfer	$\frac{\text{Cal.}}{\text{m.}^2 ^\circ \text{C. sec.}}$
v	Velocity, y direction	$\frac{\text{m.}}{\text{sec.}}$
w	Relative vapor velocity	$\frac{\text{m.}}{\text{sec.}}$
y_0	Condensate thickness	m.
z	Viscosity	$\frac{\text{kg. sec.}}{\text{m.}^2}$
θ	Temperature difference	$^\circ \text{C.}$
ϕ	Angle	radians
ρ	Density	$\frac{\text{kg.}}{\text{m.}^3}$

If the correct units were used, the numerical value of the criterion would be independent of the particular system used. It so happens that in the technical system Reynolds' criterion must be $Du\rho/2g$ in order to be dimensionless.

A large part of this paper deals with the theoretical work of Nusselt, who used the older technical units throughout. In order to use Nusselt's derivations in the most simple manner, it is necessary to adopt a technical English system that is comparable with the system used by him, and, most important of all, a system in which the fundamental units are consistent.

Throughout this discussion the old technical metric system is used for fundamental calculations. The final results of heat-transfer coefficients, etc., are calculated on the hour basis by multiplying by the factor 3600. To convert to the English system, it is merely necessary to divide the value thus obtained by 4.88.

Experimental and Theoretical Results to Date

Probably Joule (27) was the first one to understand clearly the fact that the major resistance to heat flow from a fluid to a solid surface is a film of nearly stagnant fluid at the interface. Both he and Lord Kelvin realized that the thickness of this film was indeterminate, since in general heat flows both by conduction and convection through the film. Kelvin suggested that, instead of "thermal conductivity," the term "heat-transfer coefficient" be used for such cases. This coefficient was defined as the amount of heat passing through unit surface in unit time with unit temperature difference.

In 1873 Reynolds (48) studied the effect of air on the over-all coefficients of heat transfer from steam to water. He concluded that there is no limit to the rate at which heat will flow from pure condensing steam to a metal surface except the power of the surface to carry away the heat. He found that a small amount of air in the steam has a very marked effect in decreasing the flow of heat.

From the time of Reynolds until about 1910 no significant experiments were made. A large number of investigators worked on the over-all coefficients of heat transfer from steam to water in condensers and evaporators, but these were largely tests on specific apparatus.

Since 1910 rather a large number of experiments have been made measuring the actual metal temperatures when steam was condensed on tubes. In this way the heat-transfer coefficients of the condensing vapor could be determined.

Webster (54), Clement and Garland (8), McAdams and Frost (33), Morris and Whitman (37), and Othmer (40) have investigated the condensation of steam on the outside of a horizontal tube. McAdams and Frost also reported experiments on benzene and carbon tetrachloride. Jakob,

Erk, and Eck (24) have studied steam condensation inside a horizontal tube.

Jordan (25), Callendar and Nicolson (6), Philipp (42), and Jakob and Erk (22) have studied steam condensation on vertical tubes. Badger, Monrad, and Diamond (4) have reported results obtained with diphenyl vapor condensing at high temperature differences on a long vertical tube.

Othmer (40), Merkel (35), Josse (26), Arzoomanian and Alpert (1), Kerr (29) and Robinson (49), Chambers and Eskew (7) and very recently Colburn and Hougen (9) have studied the effect of air on the condensation of steam.

The effects of vapor velocity have been studied experimentally by several of the above investigators, but no very definite conclusions were reached. The effect of superheat of the vapor has been studied by Jakob and Erk (22) and shown to have a very small effect on the transfer of heat.

In 1916 Nusselt (38) in an excellent mathematical paper, studied the effects of vapor and liquid properties, shape of the surface, and the purity and superheat of the vapor on the condensation of any vapor on a solid surface. This paper has not received the attention it merits, possibly because it has not been abstracted in this country.

Note—The authors have found that a large amount of foreign literature on such subjects as heat transfer, fluid motion, diffusion, etc., has not been abstracted in English. Considerable advances in these fields have been made in Germany during the past twenty years that have never been noticed by the abstracting journals. It is especially to be noted that only very recently *Chemical Abstracts* has been covering these fields to any extent and thus should not be relied upon solely. Some additional information may be obtained in the *Engineering Index* and the *Industrial Arts Index*, but these subjects are covered most completely in the *Technische Zeitschriftenschau*.

This theory will be reviewed in some detail as it is of fundamental importance. Application of this theory to experimental work of the authors and others will be shown.

In 1925 Stender (52) applied Nusselt's theory to superheated steam and calculated several formulas for this case. Merkel (34), Gröber (15) and ten Bosch (5) give excellent summaries of Nusselt's work on the condensation of vapors.

In 1920 Parr (41) calculated steam-film coefficients on somewhat different assumptions than Nusselt but using the same general idea. His assumptions did not hold for conditions actually found in practice as well as those of Nusselt; and the treatment was not nearly so thorough. A brief comparison of the two theories is given by Jakob (21).

Nusselt's Theory of Condensation of Vapors

CASES CONSIDERED AND FUNDAMENTAL ASSUMPTIONS—
Nusselt considered the following five fundamental cases:

(1) Vapor condensing on a smooth, plane surface making the angle ϕ with the horizontal, assuming that the vapor is pure and saturated and relatively stationary with respect to the condensate.

(2) Vapor condensing on the outside of a horizontal tube under the above conditions.

(3) Vapor condensing on surface as in (1) but with appreciable vapor velocity.

(4) Superheated vapor condensing on any surface.

(5) Vapor condensing on any surface, vapor impure.

Nusselt first pointed out that surface tension of the condensate can have no effect on the thickness of film formed, and this thickness may be calculated from hydrodynamical considerations.

In order to simplify the mathematical treatment, Nusselt made several assumptions, some of which he was later able to prove. These assumptions should be clearly borne in mind throughout the discussion and are essentially as follows:

(1) The film of condensate is so thin that the temperature gradient through it is a straight line.

(2) The heat is all carried to the metal surface by pure conduction in the direction perpendicular to the surface.

(3) Physical properties of the condensate may be taken at the mean film temperature.

(4) The surface is relatively smooth and clean.

(5) The film of condensate always moves in viscous motion.

(6) The curvature of the film may be neglected.

(7) The temperature of the solid surface is constant.

CASE 1. VAPOR CONDENSING ON INCLINED TUBE OR WALL—Consider a wall making the angle ϕ with the horizontal and kept at a uniform temperature t_w . Saturated pure vapor at a temperature t_v is condensing on this surface. Let x be the direction parallel to the wall and y the direction perpendicular to it—i. e., through the condensate film.

At any point in the condensed film a shear stress, S , is set up owing to the viscosity of the condensate and the variation of velocity of the condensate with distance from the wall.

$$S = z \frac{du}{dy}$$

and
$$dS = z \frac{d^2u}{dy^2} dy$$

If ρ is the density of the condensate, the differential shear stress set up as above must be counterbalanced by the differential weight of the condensate tending to pull the condensate downward, or

$$dS + \rho \sin \phi dy = 0$$

and
$$\frac{d^2u}{dy^2} = - \frac{\rho}{z} \sin \phi$$

This integrates at once to

$$u = -\frac{\rho \sin \phi y^2}{2z} + C_1 y + C_2$$

Since at $y = 0$, $u = 0$; therefore $C_2 = 0$, and if the vapor is stationary there is no force tending to shear the condensate at the surface and

$$z \frac{du}{dy} = 0, \text{ at } y = y_0$$

Thus

$$\frac{du}{dy} = 0$$

and

$$C_1 = \frac{\rho \sin \phi y_0}{z}$$

Therefore

$$u = \frac{\rho \sin \phi}{2z} (2y y_0 - y^2)$$

The mean condensate velocity, u_m , at any point down the wall x is given by the following equation:

$$u_m = \frac{1}{y_0} \int_0^{y_0} u dy = \frac{\rho}{3z} y_0^2 \sin \phi$$

If k is the thermal conductivity of the condensate, then the heat passing through unit surface in unit time at the point x is

$$Q = \frac{k}{y_0} (t_v - t_w)$$

If L is the latent heat and G the weight of vapor condensed, then

$$G = \frac{k}{L y_0} (t_v - t_w)$$

represents the amount of vapor condensed on unit surface in unit time.

For simplicity consider a vertical wall of width B and height H . In a unit time the amount of condensate flowing past a point x down the wall is

$$\rho u_m y_0 B = \frac{\rho^2 B y_0^3}{3z}$$

At dx lower

$$\rho B d(u_m y_0) = \frac{\rho^2 B y_0^2 dy_0}{z}$$

more condensate flows downward. This differential amount of condensate must be due to the vapor condensed in the strip $B dx$, or

$$dG = \frac{k(t_v - t_w) B dx}{L y_0}$$

or

$$\frac{k(t_v - t_w) dx}{L y_0} = \frac{\rho^2 y_0^2 dy_0}{z}$$

(7)

Therefore

$$x = \frac{L\rho^2 y^4}{4kz(t_v - t_w)} + C$$

Since at $x = 0$, no liquid runs down from above, $C = 0$, and

$$y_0 = \sqrt[4]{\frac{4kz(t_v - t_w)x}{L\rho^2}}$$

Thus Q at the point $x =$

$$Q_x = \sqrt[4]{\frac{L\rho^2 k^3 (t_v - t_w)^3}{4zx}}$$

If the heat-transfer coefficient, U_v , is defined by the equation

$$Q_x = U_v(t_v - t_w)$$

then

$$U_v = \sqrt[4]{\frac{L\rho^2 k^3}{4zx(t_v - t_w)}} \text{ at the point } x$$

To obtain a mean coefficient it is necessary to integrate Q_x over the length H and the width B .

$$\text{Therefore } BQH = B \int_0^H Q_x dx = \frac{4}{3} B \sqrt[4]{\frac{L\rho^2 k^3 H^3 (t_v - t_w)^3}{4z}}$$

$$\text{or } U'_{mv} = \frac{4}{3} \sqrt[4]{\frac{L\rho^2 k^3}{4zH(t_v - t_w)}} = \frac{0.943 \sqrt[4]{A}}{\sqrt[4]{H(t_v - t_w)}} \quad (1)$$

where $A = L\rho^2 k^3 / z$ and is a function of the condensate properties alone. It may be plotted versus temperature for any fluid.

If the wall is at the angle ϕ with the horizontal

$$U'_{mv} = 0.943 \sqrt[4]{\frac{L\rho^2 k^3 \sin \phi}{zH(t_v - t_w)}} \quad (1a)$$

Nusselt showed that from the equation of continuity there is a component of condensate velocity toward the wall, and therefore some heat is carried by convection to a point nearer the wall. This velocity may be calculated and the heat carried in this manner is found to be negligible. The error caused by neglecting this effect is doubtless no greater than that due to the assumption of a straight-line temperature gradient through the film.

Figure 1 shows Nusselt's theory applied to saturated steam condensing on a vertical wall 1 meter high. The vapor temperature is 100°C ., and the wall 90°C . The condensate thickness at any point is shown by the curve starting at $x = 0$. At four points on the wall the velocity components of the condensate in the x and y direction are given by the curves labeled u and v , the scales of which are indicated at the right. Four stream lines are shown labeled as $\psi = \text{constant}$, and these indicate the path of the condensate as it moves down the tube. It is apparent that condensation is very rapid at the top of the wall and gradually becomes less.

Nusselt pointed out that, since there is a temperature gradient in the film, some sensible heat is given up by the condensate and the above calculation is somewhat in error. The error is never more than 1 to 3 per cent even at very high temperature differences. He has developed a more exact formula which is of very little practical significance.

CASE 2. VAPOR CONDENSING ON HORIZONTAL TUBE—For horizontal tubes the same assumptions are made as above, but it is necessary to consider the force of gravity acting in the direction tangential to the tube surface. Nusselt found that the average coefficient over the whole tube may be given as

$$U_{mv} = 0.8024 \sqrt[4]{\frac{2L\rho^2k^3}{3zD(t_v - t_w)}} = \frac{0.725\sqrt[4]{A}}{\sqrt[4]{D(t_v - t_w)}} \quad (2)$$

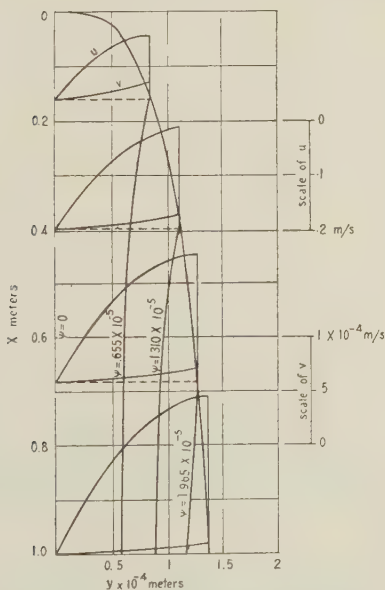


Figure 1—Saturated Steam Condensing on Wall 1 Meter High (Nusselt)

where D is the diameter of the tube.

Note—Colburn and Hougen (9) have mentioned this equation and calculated the constant to be 0.61 instead of 0.725. This was apparently due to the omission of the fourth root of 2.

$$\left(0.8024 \sqrt[4]{\frac{1}{3}} = 0.61, \text{ but } 0.8024 \sqrt[4]{\frac{2}{3}} = 0.725 \right)$$

The upper half of the tube condenses 59.4 per cent of the vapor if no condensate flows from tubes above. If tubes are in banks and the condensate from one drips on the one below, the coefficient is decreased. For the case of a tube directly below a single tube the constant in Equation

2 becomes 0.545, and becomes still less for tubes lower in the bank.

Nusselt applied Equation 2 to the experiments of English and Donkin (13) and found fair agreement in the case of smooth tubes, although these experiments were made on very crude apparatus and could not be expected to give very accurate results. The agreement between theory and experiment was of the order of magnitude of ± 20 per cent.

CASE 3. EFFECT OF VAPOR VELOCITY—If vapor is moving with a velocity w with respect to the outside of the condensate film on a vertical wall or tube, the constant C_1 can no longer be determined from the condition that there is no shearing force at the surface, since the vapor exerts a force S per unit area on the condensate surface.

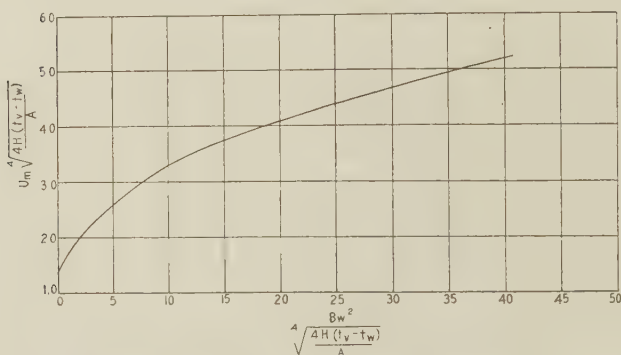


Figure 2—Effect of Vapor Velocity (Ten Bosch)

For the case of steam flowing through a tube or pipe Nusselt was able to calculate the constant C_1 by considering the pressure drop through the tube. The experiments of Eberle (11) showed that

$$\frac{dP}{dl} = \frac{C_3 u_v^2 \rho_v}{D}$$

where $C_3 = 10.5 \times 10^{-4}$ sec./m.

The subscript v refers to the vapor.

Since the friction forces on the tube circumference must equal the pressure forces on the tube cross section,

$$S\pi D = \frac{\pi D^2}{4} \frac{dP}{dl}$$

and

$$S = \frac{C_3 w^2 \rho_v}{4} \text{ if } w = u_v$$

Therefore

$$S = \frac{\pm C_3 w^2 \rho_v}{4} = z \frac{du}{dy} = -\rho y_0 \sin \phi + C_1 z$$

(10)

or

$$C_1 = \frac{\pm C_3 w^2 \rho_v}{4z} + \frac{\rho y_0}{z} \sin \phi$$

If the steam velocity is in the direction of the condensate drainage the positive sign is used, and the negative if the flows are countercurrent.

Proceeding in general as with stationary vapor, Nusselt was able to show that

$$\frac{U_{mv}}{k} \sqrt[4]{\frac{H}{a}} = \text{function of } \sqrt[4]{\frac{b}{a}}$$

where

$$a = \frac{L\rho^2}{4kz(t_v - t_w)}$$

and

$$b = \frac{C_3 w^2 \rho_v}{3\rho}$$

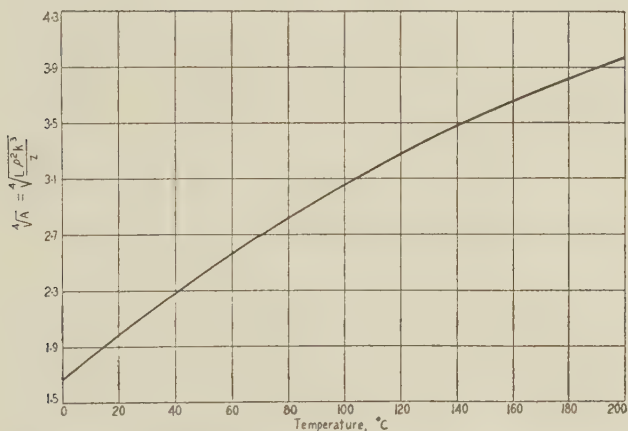


Figure 3— $\sqrt[4]{\frac{L\rho^2 k^3}{z}}$ for Water (Old Technical Metric Units)

Ten Bosch (5) has pointed out that if the above relationship is true, then

$$U_{mv} \sqrt[4]{\frac{4H}{A} (t_v - t_w)} = \text{function of } \sqrt[4]{\frac{Ew^2}{A} (t_v - t_w)}$$

where

$$A = \frac{L\rho^2 k^3}{z}$$

and

$$E = \frac{b}{kw^2} = \frac{C_3 \rho_v}{3\rho k}$$

This function is shown in Figure 2 for concurrent flow of vapor and condensate. The value of $\sqrt[4]{A}$ is shown for water in Figure 3 in metric units. (L may be taken at mean film temperature for small temperature differences.)

Nusselt discussed several very interesting cases for the flow of vapor countercurrent to condensate drainage and showed that sometimes all or part of the condensate is carried to the top of the wall. It is necessary to use very

high vapor velocities in order to do this, and such cases would seldom be found in commercial practice.

Nusselt did not attempt to apply the theory described above to any other vapor than steam and only then in the case of steam condensing inside of a tube since the constant C_3 was unknown in all other cases. This is discussed later, and a method shown by means of which the effect of velocity of any vapor in any cross section may be calculated.

CASE 4. EFFECT OF SUPERHEAT—There are two sub-cases to the case of heat transfer from superheated vapor. When the wall temperature is above that of saturated vapor at the given pressure, no vapor will condense and the heat transfer will be the usual one from a gas or vapor to a solid surface. When the wall temperature is less than that of saturated vapor, condensation will take place, but the phenomenon is not the same as in the case of saturated vapor. There is a temperature difference between the vapor and the surface of the condensate corresponding to the degrees of superheat. Heat is carried to this interface by the usual gas-solid method. Thus if U_s is the heat-transfer coefficient from vapor to interface,

$$U_s(t_s - t_w)dx + \frac{L\rho^2 y_0^2 dy_0}{z} = \frac{k}{y_0} (t_v - t_w)dx$$

$$\text{or} \quad \frac{dy_0}{dx} = \frac{kz(t_v - t_w)}{L\rho^2 y_0^3} \left[1 - U_s \frac{(t_s - t_v)y_0}{k(t_v - t_w)} \right]$$

$$\text{If} \quad h = \frac{U_s(t_s - t_v)}{k(t_v - t_w)}$$

$$\text{then} \quad x = \frac{L\rho^2}{kz(t_v - t_w)} \left[\frac{1}{h^4} \log \frac{1}{1 - hy_0} - \frac{y_0^3}{3h} - \frac{y_0^2}{2h^2} - \frac{y_0}{h^3} \right]$$

Since $hy_0 < 1$, the logarithm may be expanded into a power series,

$$\text{or} \quad x = \frac{L\rho^2}{kz(t_v - t_w)} \left[\frac{y_0^4}{4} + \frac{hy_0^5}{5} + \frac{h^2 y_0^6}{6} + \frac{h^3 y_0^7}{7} + \dots \right]$$

Note—In the original article Nusselt had $-y_0^3/h^3$ and y_0^4/h instead of $-y_0/h^3$ and $y_0^4/4$. This was probably due to a typographical error. Obviously if h becomes zero (no superheat), the equations should degenerate into the one previously calculated for saturated steam.

This complicated power series is not satisfactory to use in practice. Recently Stender (52) and Merkel (34) have shown that a very good approximation may be made, since the amount of heat flowing is proportional to the condensate thickness and to the cube root of the amount of condensate if t_v and t_w are the same. Therefore

$$\frac{Q_s}{Q} = \frac{y_0}{y_{0s}} = \sqrt[3]{\frac{G}{G_s}} = \beta$$

where the values with the subscript s refer to the condensation of superheated vapor and those without subscript to the saturated vapor. β has been shown by Merkel to be a

function of $\frac{U_s(t_s - t_v)}{U(t_v - t_w)}$ alone and may be plotted as in Figure 4. U_s must be calculated as the usual heat-transfer coefficient from a superheated vapor or gas to a wall, and depends on the velocity, temperature, etc.

For an example, consider the condensation of saturated steam on a vertical wall 0.5 meter high; vapor at 100° C.; temperature difference to wall, 5° C. The mean film temperature = 97.5° C., and therefore from Figure 3

$$U = \frac{0.943}{\sqrt{0.5 \times 5}} \times 3.03 \times 3600 = 8190 \text{ Cal./m}^2\text{/hr./}^\circ\text{C.}$$

Let the vapor be superheated to 200° C. Then $t_s - t_v = 100^\circ\text{C.}$, and $t_v - t_w = 5^\circ\text{C.}$ Assume that $U_s = 50$. Then

$$\frac{U_s(t_s - t_v)}{U(t_v - t_w)} = 0.1220$$

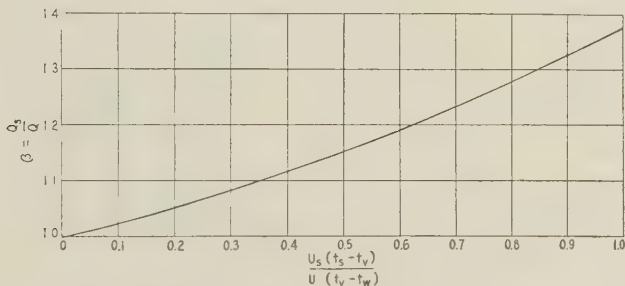


Figure 4—Effect of Superheat (Merkel and Stender)

and from Figure 4 the ratio of the amount of heat flowing to that with saturated steam is 1.032. Thus, there is only a 3 per cent increase in the amount of heat, which is nearly negligible. If the steam velocity is increased, the value of U_s is increased and the effect of superheat is greater.

CASE 5. EFFECT OF NON-CONDENSABLE GASES—The effect of non-condensable gases on the heat-transfer coefficient is very important technically. However, exact calculation of the effect is very difficult and Nusselt merely indicated a general method. Colburn and Hougen have recently attacked this problem theoretically and experimentally with some definite results.

Applications of Nusselt's Theory

CASE 2. VAPOR CONDENSING ON HORIZONTAL TUBE—The condensation of pure saturated vapors on the outside of a horizontal tube has been studied experimentally by McAdams and Frost (35), Morris and Whitman (37), Clement and Garland (8), Othmer (40), Webster (54) and several others.

Figure 5 shows Nusselt's Equation 2 applied to Othmer's data for approximately pure steam. The vapor velocity

was in all cases very small and its effect should be negligible in all of Othmer's work. This is apparently true, as the experimental data check the theory almost within the experimental error. For high temperature differences Othmer's data spread for different steam temperatures and do not follow the theoretical curves very well. It seems scarcely plausible that a change of only 15° C. in the temperature of the steam could have as great an effect on the coefficient as

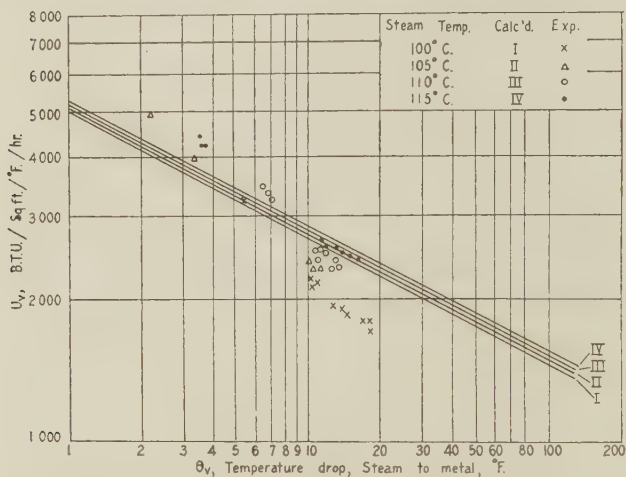


Figure 5—Application of Equation 2 to Othmer's Data

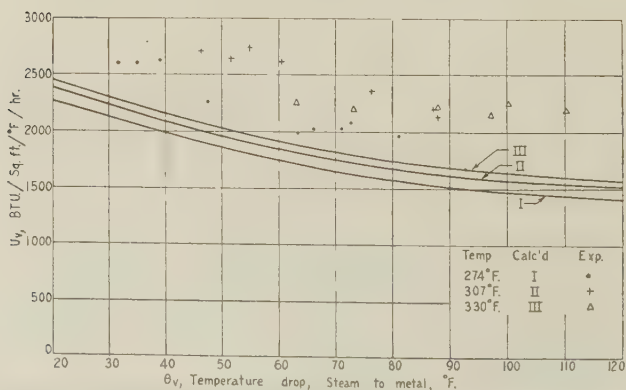


Figure 6—Application of Equation 2 to Clement and Garland's Data

that indicated by the data. The experimental difficulties in obtaining pure steam probably account for most of the discrepancy between theory and experiment.

Of even greater interest, perhaps, is the application of the theory to the data of McAdams and Frost on steam, benzene, and carbon tetrachloride, as shown in Table I. The experimental data on steam check the theoretical values more closely than those of Othmer's except for one point,

which is probably too high. The checks on carbon tetrachloride and benzene are especially interesting, as no satisfactory explanation of the data has appeared previously. McAdams and Frost attempted a rough correlation by assuming that the coefficient increased with the first power of the thermal conductivity and fluidity of the condensate. This was admittedly empirical and very unsatisfactory for the prediction of coefficients of even the same vapors under

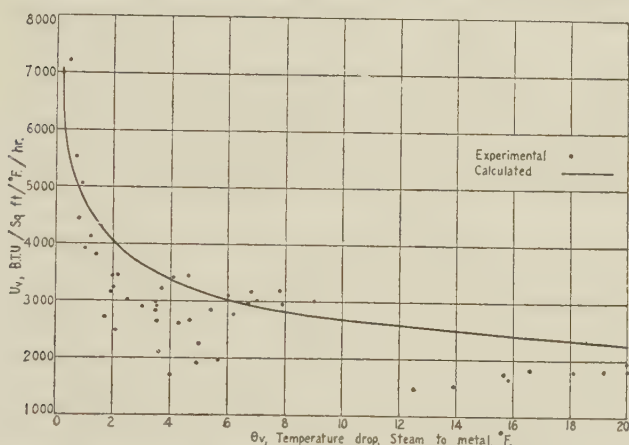


Figure 7—Application of Equation 2 to Morris and Whitman's Data

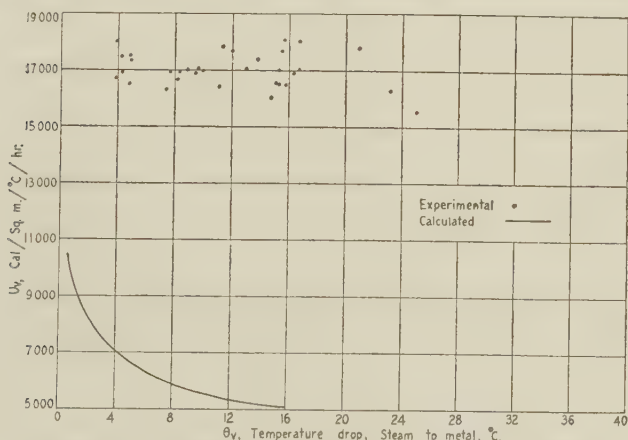


Figure 8—Application of Equation 1 to Philipp's Data

different conditions. Actually the coefficient varies as the three-quarter power of the thermal conductivity and as the fourth root of the fluidity. In addition many other important factors, such as density and latent heat of the liquid and the size and shape of the tube, have a great effect.

Clement and Garland had an indefinite amount of steam blowing out continuously in their experiments. This probably accounts for the fact that their coefficients are about

30 per cent above those calculated by Nusselt's equation. (Figure 6)

Table I—Nusselt's Theory Applied to Data of McAdams and Frost
(Tube 0.675 inch o. d. = 0.01715 meter)

RUN	AV. VAPOR TEMP. ° C.	AV. TEMP. DIFF. ° C.	APPROX.	U	
			FILM TEMP. ° C.	EXPT. <i>B. t. u./sq. ft./° F./hr.</i>	CALCD.
H ₂ O	99.3	19.3	89	2190	2075
H ₂ O	102.3	20.6	90	2110	2045
H ₂ O	105.1	23.7	93	2060	1990
H ₂ O	100.5	22.1	89	2150	1995
H ₂ O	116.0	12.0	110	3360	2535
H ₂ O	99.7	13.4	94	2310	2319
H ₂ O	99.1	12.4	93	2475	2360
CCl ₄ ^a	75.1	14.8	67	283	303
Benzene ^a	78.4	15.5	69	306	375
Benzene ^a	78.9	12.6	72	366	396

^a It was found impossible to obtain accurate values for the thermal conductivities of carbon tetrachloride and benzene, as no two investigators agree within 20 per cent. The values used by McAdams and Frost were used here as they appeared to be the average values in the literature.

The effect of vapor velocity outside a tube and at right angles to the condensate drainage was not calculated by Nusselt. It is apparent that the calculation would be extremely tedious in such a complicated case and the results would be doubtful.

Webster's data afford a good experimental basis on which to calculate the effect of vapor flow at right angles to the condensate flow. His data are somewhat higher than those calculated from Equation 2, because of the high velocities used. Very few practical applications will be found in which such high velocities are used.

Morris and Whitman heated oils with steam and it was possible to calculate the steam-film coefficient from their data. Since in most cases their temperature differences between steam and tube surface were low, the coefficients obtained from their data are probably not very accurate. In general their data were about 20 per cent lower than the theoretical. (Figure 7)

For the case of vapors condensing on the outside of horizontal tubes, therefore, it appears that the assumptions made by Nusselt are quite well fulfilled in practice under ordinary conditions if proper venting of non-condensable gases is insured. However, the vapor must be relatively stationary and the tube surface smooth and clean. Under these conditions the coefficient of heat transfer may be calculated for any vapor within a probable accuracy of 10 per cent, if the physical properties of the vapor are known.

For vapors condensing inside horizontal tubes no theoretical equation has been developed. Removal of the condensate is by means of a slight inclination of the tube and the use of high vapor velocities. The work of Jakob, Erk, and Eck gives experimental data for this case that may be used with proper caution for the prediction of coefficients.

CASES 1 AND 3. STATIONARY OR MOVING VAPOR CONDENSING ON INCLINED WALL—Not many data are available to check the theory for the case of vapors condensing on a

tube or a vertical or inclined wall. The data of Philipp (42), Callendar and Nicolson (6), Jordan (25), Jakob and Erk (22) on steam, and of Badger, Monrad, and Diamond (4) on diphenyl vapor are the only ones that the authors have found that may be relied upon.

Philipp's data were rather erratic, and in only one series of runs was it possible to obtain consistent results. The coefficient obtained by him for steam at 99° C. condensing on a 4-foot tube was about 3480 in English units. The 6- and 8-foot sections gave him about the same value independent of temperature difference. The steam velocities used by him were very low (less than 30 feet per second), and according to Nusselt's theory should have had a negligible effect. In several series of runs he found an increase in coefficient with temperature difference. Figure 8 shows

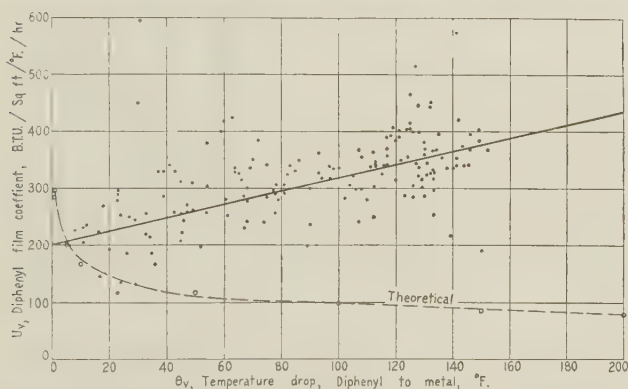


Figure 9—Application of Equation 1 to Condensation of Diphenyl

his experimental results at 99° C. and Nusselt's theoretical curve (Equation 1) for his 4-foot length. It is apparent from the curve that Nusselt's theory does not hold and gives values that are much too low at high temperature differences.

Callendar and Nicolson's experiments on a 2-foot vertical thick-walled tube gave coefficients roughly about 2700 when a temperature difference of about 30° F. was used. According to Nusselt the coefficient should be

$$U_v = \frac{0.943 \sqrt[3]{A}}{\sqrt[3]{H(t_o - t_w)}} = \frac{0.943 \times 2.97 \times 3600}{\sqrt[3]{0.609 \times 16.7}} = 5270 \text{ (new metric) or } = 1169 \text{ (English)}$$

Thus the experimental values are more than twice the theoretical based on zero steam velocity. No data were given concerning the velocities, but they were probably not very high.

Results were obtained in a 1/4-inch diameter platinum tube that were at least twice the theoretical, even when the flow of steam was countercurrent to condensate drainage;

in fact, even higher results were obtained in the latter case. It appeared to Callendar and Nicolson that when steam is condensing on the outside of very small tubes moving drops of water are formed which aid rather than retard the flow of heat, owing possibly to the increased surface.

In the experiments on diphenyl by Badger, Monrad, and Diamond (4) the vapor coefficient was found to increase markedly with temperature difference and to be independent of temperature, whereas the theory gave a curve that showed a rapid decrease in coefficient with temperature difference. From Figure 9 it is obvious that the vapor velocity is not sufficient to explain the fivefold increase in coefficient. This will be shown later.

Jakob and Erk found that for their short tube (0.466 meter) Nusselt's theory gave rather fair results, but not by any means exact. In all cases the theoretical results were high, whereas in other work on longer vertical tubes the theory gives low results. Figures 10 and 11 show the results obtained by Jakob and Erk with saturated and superheated steam. The data are shown in solid lines which represent the amount of heat per square meter per hour, and thus must be divided by the temperature difference to obtain the heat-transfer coefficient. With wall temperatures above the condensation temperature the coefficient for superheated steam was, of course, the usual one from a gas to a solid. Some previous work by Poensgen (43) on heat flow from superheated steam is shown in dotted lines.

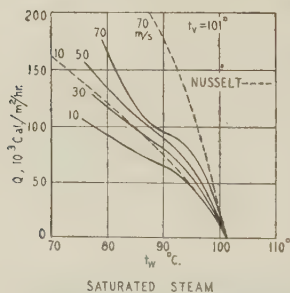


Figure 10—Condensation of Saturated Steam from Data of Jakob and Erk

Jordan's data on a vertical tube 1 meter long are given in Figure 12 and Nusselt's theoretical curves are also plotted. It is apparent that the data check the theory qualitatively at low vapor velocities, but not at high. At extremely low velocities the theoretical results are high. This agrees to some extent with the conclusions of Jakob and Erk.

Apparently, therefore, one or more of the assumptions made by Nusselt does not hold for rather long vertical tubes when temperature differences of any magnitude are used. According to Nusselt the use of a horizontal tube is preferable, especially when the tubes are long and high temperature differences are used, but experiment tends to show that vertical tubes are to be preferred under these conditions.

Limitations and Corrections of Nusselt's Theory

Whether it is permissible to use the mean film temperature for fluid properties is not clear at present. At high

temperature differences and with fluids whose properties change very markedly with temperature, it may be better to recalculate Nusselt's equations using these properties as variables. However, the assumption of a linear temperature gradient in the film as well as that of a smooth surface probably leads to as much as, if not more error than, that of mean fluid temperature.

EFFECT OF VAPOR VELOCITY INSIDE CIRCULAR TUBE—One point in Nusselt's derivation was found to be obviously in error. In calculating the pressure drop due to steam flowing through a circular tube, the use of some equation such as the following has been shown to be exact for all fluids:

$$\frac{dP}{dl} = \frac{2f\rho u^2}{gD}$$

where f = friction factor
 ρ = fluid density
 u = fluid velocity
 g = acceleration of gravity
 D = tube diameter
 P = pressure
 l = tube length

The friction factor, f , has been shown to be a function of Reynolds' criterion, which is the dimensionless group Du/ν . f is the same for the same values of this criterion, and may be obtained quite exactly from Jakob and Erk's (20) equation

$$f = 0.00714 + 0.610 Re^{-0.35}$$

where Re is the Reynolds' criterion calculated on the diameter of the tube. It is obvious that Nusselt's constant C_3 is equal to $f/2g$, and is, accordingly, not a constant, but a function of Reynolds' criterion. Moreover, this calculation of C_3 is valid, not only for steam, but for all fluids.

The original experiments by Eberle (11) on the determination of the drop in pressure of steam through a tube were made in 1908. He used a combination of pipes joined together with ten flanges and three right-angled bends. In addition to this, every time a high velocity was used the corresponding pressure of the steam was low, and the value of Reynolds' criterion was approximately constant, since the viscosity of the vapor is fairly constant. Naturally, the friction factor was found to be even more constant, as it varies approximately with the fourth root of Re . Eberle stated that previous work had shown the constant to be as high as 15×10^{-4} second per meter, but that his experiments showed conclusively that the value was about 10.5×10^{-4} . In all probability both experimenters were correct since C_3 is not a constant.

It is apparent that the so-called effect of vapor velocity on the heat-transfer coefficient is really the effect of frictional forces on the surface of the condensate. The factor "velocity" is rather a poor one, since with the same velocity

the frictional forces may be quite different. Thus a certain velocity inside a tube has a much different effect from the same velocity outside a tube, or in fact in a different tube.

EFFECT OF VAPOR VELOCITY IN CROSS SECTIONS OTHER THAN CIRCULAR—For vapor condensing inside a circular tube the above method of the calculation of the effect of velocity is valid, provided all the assumptions made are true. With the vapor condensing outside a circular tube, or in fact in any other cross section, the problem is quite different. It has been shown (2, 20, 50, 14, 18) that for a first approximation the pressure drop through any cross section may be calculated from the following equation if the fluid is in turbulent motion:

$$\frac{dP}{dl} = \frac{f\rho u^2}{2gm}$$

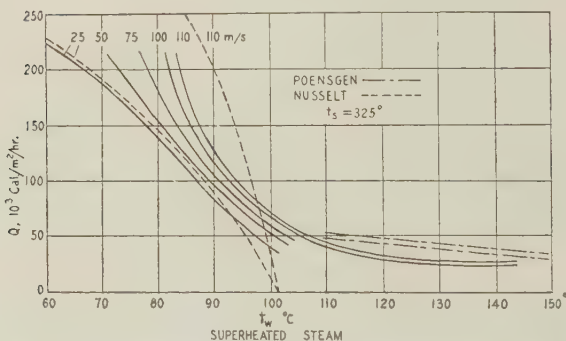


Figure 11—Condensation of Superheated Steam from Data of Jakob and Erk

where m is the so-called hydraulic radius which is defined as the ratio of the free cross-sectional area divided by the wetted perimeter. Thus in a circular tube the value of m is $1/4 D$. Many cross sections have been used—circular, annular, star, rectangular, open channels, etc.—and the results have been fairly consistent. (Figure 13)

Assuming that the concept of hydraulic radius is correct for concentric tubes, let us proceed to find the effect of vapor velocity on the film of condensate forming on the outside of the inner tube. This is the condition found in the work presented in the experiments of Badger, Monrad, and Diamond on diphenyl vapor and, in fact, in most vertical heating surfaces.

Let r_1 and r_2 be the radii of the inner and outer tubes, respectively, and s_1 and s_2 the respective frictional forces per unit area on the two surfaces. Then,

$$2\pi s_1 r_1 + 2\pi s_2 r_2 = \frac{dP}{dl} \left(\frac{r_1^2 - r_2^2}{2} \right) \pi$$

The hydraulic radius

$$m = \frac{(r_1^2 - r_2^2)\pi}{2\pi(r_1 + r_2)} = \frac{r_1 - r_2}{2}$$

and $S = \frac{dP}{dl} \times m$, if $s_1 = s_2$ and $S = \frac{f\rho_v u_v^2}{2g}$

If s_1 and s_2 are the same, it is easy to calculate the force on the outer part of the condensate. There is nothing to show that they should be alike, but it is very probable that they are if the concept of hydraulic radius holds. Doubtless they are of the same order of magnitude, at least, and some idea of the effect of vapor velocity may be obtained.

The effect of vapor velocity on the coefficients reported on diphenyl were calculated on the above assumption. In the case showing the greatest effect, the coefficient was raised about 9 per cent, whereas the actual difference to be accounted for was about 400 per cent. Thus it is apparent that the effect of vapor velocity alone is not enough to account for the discrepancy between theory and experiment.

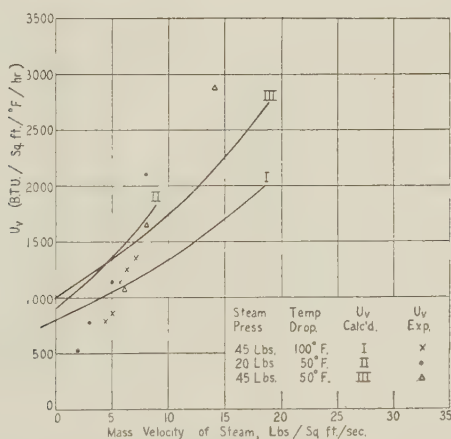


Figure 12—Effect of Vapor Velocity. Jordan's Data

EFFECT OF TURBULENCE IN FILM—Of much greater importance than the effect of vapor velocity is the effect of turbulence in the fluid film of condensate. Apparently no one has considered the possibility of turbulence except at the end of very long tubes. The authors have derived an approximate equation for the calculation of the point at which turbulence commences, and have shown that the observed deviations from Nusselt's theory may be explained by the phenomenon of turbulence.

A large number of attempts have been made to determine theoretically the start of turbulence between two parallel infinite planes when a fluid is placed between them of thickness d and the planes move with a velocity u , with respect to one another. Reynolds (47), Lorentz (30), Rayleigh (45, 46), Kelvin (28), Orr (39), Sommerfeld (51), von Mises

(36), Hopf (19), and more recently Heisenberg (17) and Lorenz (31) have studied this question mathematically. It is obviously impossible to make an experimental study of this case except very approximately owing to edge effects.

Nearly all investigators agree that turbulence should start at some value of Reynolds' criterion. Very little agreement is found in the value of the critical criterion except in the case of the work of Lorenz and Heisenberg. These two men give the value $du\rho/zg = 1560$ (approximately), and determined it by two different methods; although Lorenz states that he cannot follow the reasoning of Heisenberg. Other workers in the field give values of the criterion from 250 to 2000, but recent work has shown that the low results are in error. Probably the value of 1560 is about correct.

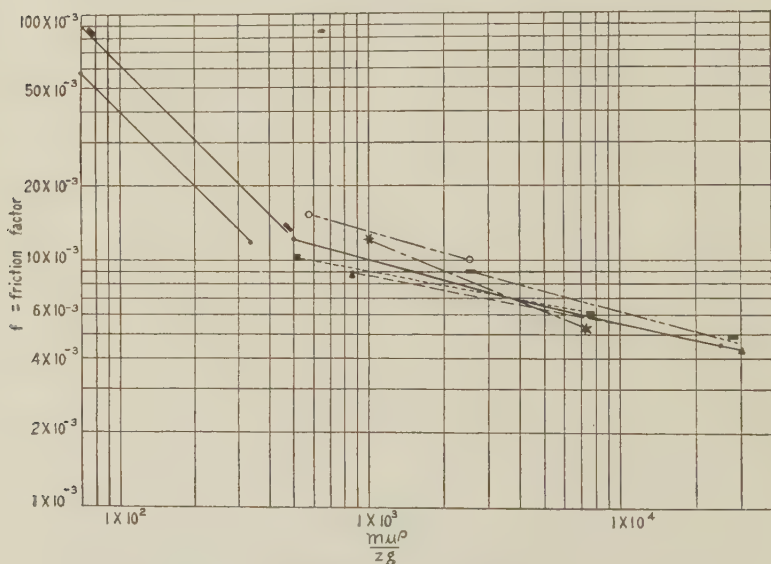


Figure 13—Friction Factor in Various Cross Sections

Couette (10), Mallock (32), Taylor (53), and Harrison (16) have studied experimentally and theoretically the critical velocity when liquid is placed between two concentric cylinders, one or both of which are rotating. It was found in all cases that instability begins at some definite value of Reynolds' criterion.

Hopf (20) points out that turbulence begins for all shapes of conduits at about the same value of Reynolds' criterion when the so-called hydraulic radius is substituted in place of D (or d). The criterion becomes $mu u_m \rho / zg$, in which u_m is the mean velocity. In most cases the critical criterion is between 300 and 400. Thus for parallel plates it is 390, for open channels 300, for rectangular channels 300 to 500, and for circular tubes 500. Since the flow of fluid down a

wall is similar to that between two plates and in an open channel, the value of the critical Reynolds' criterion was assumed to be 350.

Nusselt's equation for the velocity at any point for the flow of condensate down a vertical tube is given by the equation

$$u = \frac{\rho}{z} \left(y_0 y - \frac{y^2}{2} \right)$$

At $y = y_0$, $u = \frac{\rho}{z} \times \frac{y_0^2}{2}$, and $u_m = 2/3 u$

Therefore in technical units

$$\frac{y_0 u_m \rho}{gz} = 350 = y_0 \frac{\rho y_0^2 \rho}{3 z gz} = \frac{\rho^2}{3 gz^2} \left[\frac{4kz x_c (t_v - t_w)}{L \rho^2} \right]^{3/4}$$

Therefore
$$x_c = \frac{2668 L z^{5/3} g^{4/3}}{\rho^{2/3} k (t_v - t_w)}$$

x_c is the distance down the tube or wall at which turbulence may be expected to begin if the vapor is relatively stationary.

For water at 1 atmosphere and 10° C. temperature drop,

$$x_c = \frac{2668 \times 9.81^{4/3} \times 540 (31 \times 10^{-6})^{5/3}}{1.7 \times 10^{-4} \times 10 \times 962^{2/3}} = 5.8 \text{ meters}$$

For diphenyl under the same conditions,

$$x_c = \frac{2668 \times 9.81^{4/3} \times 140 (27.4 \times 10^{-6})^{5/3}}{1.8 \times 2.72 \times 10^{-5} \times 10 \times 840^{2/3}} = 4.4 \text{ meters}$$

For the tube used by Jakob and Erk (0.466 meter) the temperature difference necessary to start turbulence for atmospheric steam is

$$\frac{5.8 \times 10}{0.466} = 124^\circ \text{ C.}$$

Note—Colburn and Hougen (9) attempted to define the point at which turbulence would start and decided that if the coefficient for condensing steam became less than 700 in English units, turbulence would commence. Since values below 700 are not found in practice, they concluded that a turbulent condition rarely exists. This conclusion is scarcely compatible with the generally observed fact that when turbulence does appear the coefficient increases markedly. An observed coefficient greater than 700 does not, therefore, necessarily indicate a non-turbulent condition.

From the preceding discussion one would expect that for stationary steam condensing on short tubes at low temperature differences Nusselt's theory would give correct results. However, if the tube length, temperature difference, or vapor velocity is increased, a point is reached at which turbulence begins, and one would expect deviations from the theory. For the usual commercial tube lengths of 2 to 4 meters and for temperature drops of over 10° to 20° C. an effect due to turbulence may be anticipated. In the case of the experiments of Badger, Monrad, and Diamond, on diphenyl vapor, turbulence may be expected to have an effect over the entire

range of temperature difference covered. Nusselt's equations can no longer be expected to be accurate and in fact may lead to very erroneous results.

Although moderate steam velocity has rather a small effect on the coefficient as given by Nusselt's theory, it may be shown that it has a marked effect on the start of turbulence, and even in the case of very short tubes turbulence may be expected to begin with as low a temperature drop as 5°C . Thus in the tube used by Jakob and Erk, increasing the vapor velocity from 0 to 70 meters per second changed the start of turbulence from 125° to 4.4°C . temperature drop.

Just what happens when turbulence begins is of great importance both theoretically and practically, but this question offers many serious difficulties. A laminar layer of fluid must still be next to the wall, and outside of this a turbulent region. The temperature drop through this thin turbulent layer may be considered negligible and the entire drop must be through the viscous film. The thickness of this film depends to some extent on the properties of the fluid, the thickness and velocity of the turbulent layer, and also on the velocity of the vapor past the outside of the condensate, though this latter effect is probably not of such great importance as when only a viscous film is present.

At the start of turbulence at the bottom of the tube most of the surface may be treated as following Nusselt's theory. However, as the turbulent layer moves upward with increasing temperature drop, more and more heat goes through the turbulent region; and one would expect that there would be more and more deviation from the theory, until the amount of heat going through the region covered by a purely viscous film is nearly negligible. From this point on, the deviation might be expected to be about constant, or at least not to change as quickly. Apparently this is what occurs in the case of diphenyl. The coefficient increases and then gradually seems to be approaching an asymptotic value. No indication of a falling off in the coefficient was observed, even at very high temperature differences (300°F .).

Note—In addition to the coefficients shown in Figure 9, several runs were made heating water with diphenyl. The heat transferred was calculated from the rise in temperature of the water as well as from the diphenyl condensed, and the coefficient was found to be from 400 to 450 at about 300°F . temperature difference, thus showing that the curve does not continue to be a straight line as given in Figure 9, but is apparently approaching an asymptotic value.

In addition to turbulence it may be possible that some of the liquid drops off before it reaches the bottom of the tube; and also that some of the condensate may form drops and run down the outside of the film. The continued increases in coefficient is, however, rather difficult to explain on the basis of these phenomena alone, though there is some experimental evidence for their occurrence in some cases.

The apparent inconsistency obtained by most observers

in experimental work with long tubes and with high temperature differences is, no doubt, due to the fact that there is no definite boundary between the viscous and turbulent layers, but that there is a gradual change from one régime to another. This has been shown to be true with turbulent flow through pipes, and one might expect that the effect would be more marked in the case of a film of condensate. Apparently there is a sort of weaving of the interface of the two types of flow, allowing more or less heat to flow at a particular point on the tube. Average results may be obtained only by a relatively large amount of experimental work.

Jakob and Erk found that at high temperature differences the tube temperature first decreased, then increased at some point down the tube, and finally decreased slightly. This may be understood quite easily if the film is in turbulent motion. For a short distance down the tube the film is entirely viscous and therefore the wall temperature decreases as the film becomes thicker. At some point turbulence begins and the viscous layer is much thinner, allowing more heat to pass, and the wall temperature increases. As the turbulent layer increases in thickness it probably has some effect on the viscous film and the wall temperature varies slightly. The fact that this phenomenon was only observed at high temperature differences or at high vapor velocities is an added confirmation of the turbulent theory.

Jakob and Erk found also that at constant vapor velocity a point was reached when the effect of temperature difference on the coefficient increased greatly. This break was quite definite and apparently occurred at lower temperature drops as the velocity of the vapor increased, and was not apparent in the runs at very low velocities. This break was approximately at the point where turbulence would be expected to start.

Conclusion

At present all that may be said is that the fundamental theory of Nusselt is doubtless sound as is proved in the excellent correlation with the condensation of vapors on the outside of horizontal tubes, where the critical velocity is probably never reached in ordinary practice. The formulas developed for the case of vertical tubes are probably accurate for short tubes and low temperature differences; but are in general inaccurate for industrial operations, owing probably to the effect of turbulence and possibly to the formation of drops. The effect of vapor velocity as calculated by the revised method of Nusselt is probably sound when the condensate is not in turbulent flow, but in general the effect cannot be accurately determined. The qualitative agreement with the theory found by Jakob and Erk on the effects of superheat indicate that probably Nusselt's calculations are correct for this case as long as the condensate film is viscous.

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Evaporation of Caustic Soda to High Concentrations by Means of Diphenyl Vapors¹

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The dehydration of caustic soda in a forced circulation evaporator has been accomplished in semi-plant scale equipment using diphenyl vapor as a heating medium. The highest concentration obtained was slightly over 98 per cent NaOH.

Heat-transfer coefficients for both condensing diphenyl and boiling caustic are reported. In all cases the major resistance to heat flow was the film of condensed diphenyl vapor. The diphenyl coefficient was found to increase with temperature difference and to be nearly independent of vapor temperature.

THE possibilities of an apparatus in which evaporation could be carried out with high-temperature vapors as a heating medium are so extensive and so obvious that no general discussion is necessary. During the winter of 1927-28 it was decided to build an experimental evaporator for study of such problems. This paper and one on the condensation of vapors by C. C. Monrad² represent the first results of this investigation.

Selection of Working Substance

Considerable thought was given to the selection of a material that could be used as a high-temperature heating

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² To appear in a subsequent issue.

medium. A substance with a high condensing temperature at ordinary pressures was desired, since the temperature control is easy if latent heat rather than sensible heat is transferred. There were available mercury, diphenyl, diphenyl oxide, naphthalene, and other hydrocarbons of high molecular weight. Mercury vapor was considered and discarded mainly because of its poisonous nature and the difficulty of holding it in experimental apparatus. The choice between diphenyl and diphenyl oxide was not easy to make, but preliminary information indicated that diphenyl oxide was not so stable as diphenyl. The low melting point of diphenyl oxide was considered an advantage, but later developments indicated that the high freezing point of diphenyl was also an advantage in some respects. Naphthalene was rejected partly because sufficient information was not available regarding its stability and partly because of its pronounced tendency to sublime. Dibenzyl proved to have very low stability. The above information, together with the fact that diphenyl was made available in quantity by the Federal Phosphorus Company, who actively coöperated through the entire problem, placed the final decision on diphenyl.

Considerable time was spent in determining the various physical and thermal properties of diphenyl. Some determinations were made, and some properties were calculated from various equations proposed in the literature. A very extensive investigation has been conducted by the Federal Phosphorus Company, and on the completion of their work all the data on the physical properties of diphenyl will be published. At present it is sufficient to say that the boiling point of diphenyl at atmospheric pressure is 492° F. and at 140 pounds gage is 750° F. Its latent heat at atmospheric pressure is 138, and at 140 pounds gage is 97 B. t. u. per pound. The specific heat of the liquid varies between 0.4 and 0.6 in the temperature range of 200° to 700° F. Its melting point is 156.6° F.

The first application that suggested itself was that of dehydrating caustic soda with a view to eliminating the expensive and inefficient caustic pot. This paper reports the results obtained from this application.

Apparatus

As there was no previous experience to act as a guide, it became necessary to design the apparatus on the basis of rather general considerations. Many difficulties were met and the apparatus had to be modified many times. In the final form, however, most of the difficulties were eliminated, so that it is believed that the application of diphenyl to high-temperature evaporation is entirely practical.

DIPHENYL BOILER—The diphenyl boiler used in the experiments is strictly a laboratory affair and does not correspond to the design of even small commercial units. The design was based on two conditions: first, that the apparatus

- A EVAPORATOR VAPORHEAD
- B DIPHENYL JACKET
- C EVAPORATING TUBE
- D DIPHENYL VAPOR LINE
- E SURFACE CONDENSER DRIP TANKS
- F SURFACE CONDENSER
- G DIPHENYL MELTING TANKS
- H DIPHENYL HEATING TUBES
- I MOTORS FOR PUMPS
- J ELECTRIC HEATERS
- K GAS BURNER
- L DIPHENYL CIRCULATING PUMP
- M DIPHENYL FLASH CHAMBER
- N DIPHENYL DRIP RECEIVERS
- O ORIFICE PLATE
- P CAUSTIC CIRCULATING PUMP
- Q CAUSTIC STORAGE TANK
- R DIPHENYL VENT LINES
- S DIPHENYL DRIP RECEIVERS EQUALIZER LINE
- T THERMOCOUPLES
- U DIPHENYL DRIP LINE
- V STEAM COILS
- W PUMP CASING VENT

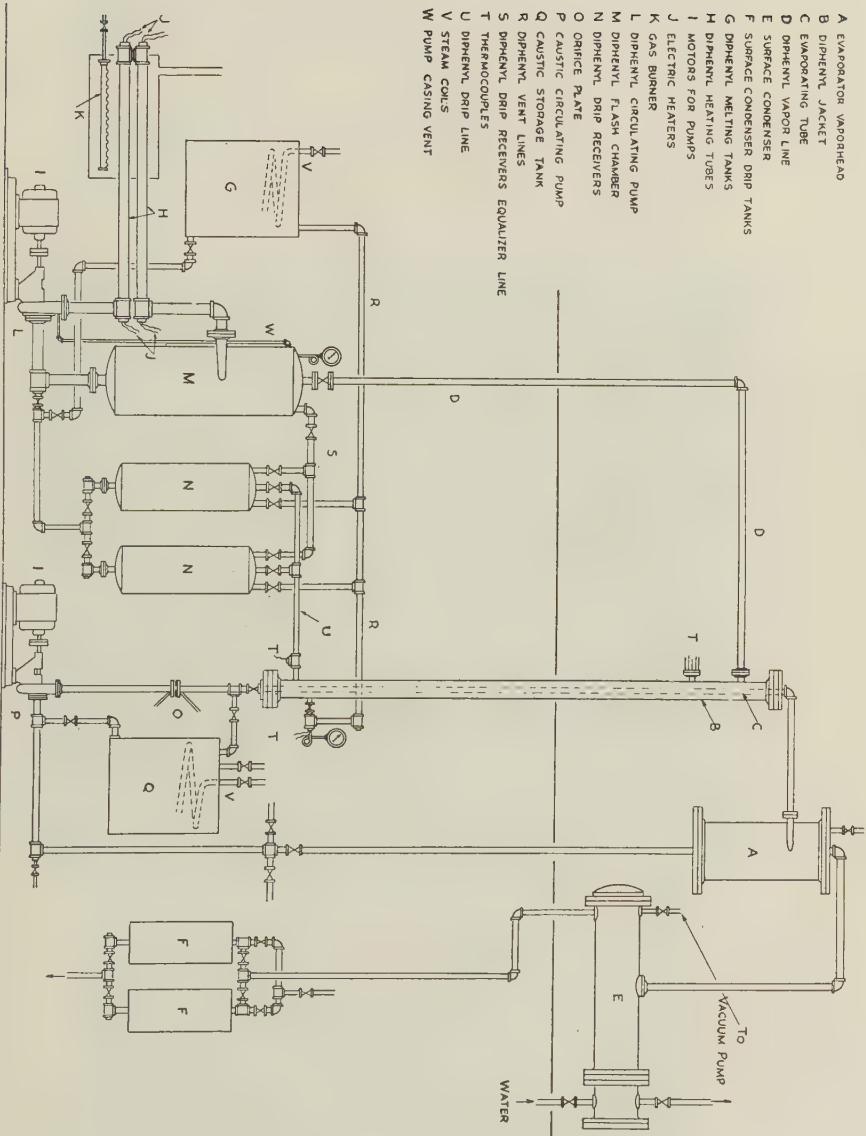


Figure 1—Experimental Evaporator and Diphenyl Boiler

should be as simple and as easy to assemble as possible; and second, that it should employ electrical heat if possible, because of ease and simplicity of control. The method adopted consisted essentially in pumping diphenyl through pipe containing electric heaters and allowing the mixture of vapor and liquid to escape tangentially into a flash chamber. The original boiler was later modified by adding gas heat to the outside of the circulating pipes.

The diphenyl boiler is shown in Figure 1. It consists of a welded steel flash chamber, *M*, 20 inches in diameter by 48 inches high. From the bottom of this flash chamber liquid diphenyl was withdrawn by a 2-inch centrifugal pump, *L*, and pumped through two pieces of 2-inch extra heavy iron pipe, *H*, each about 70 inches long, connected with return bends as indicated. Into this pipe were inserted four 6-kilowatt Chromalox immersion heaters, *J*. From the end of this pipe the mixture of liquid and vapor was conducted by a tangential inlet into the flash chamber. Gas burners, *K*, were added below the two pieces of 2-inch pipe. The diphenyl was melted in an external melting tank, *G*, with steam coils, *V*, and all vents, *R*, from all parts of the apparatus were conducted to this melting tank. The pump casing was vented to the top of the flash chamber by a separate vent line, *W*, independent of all other vents.

EVAPORATOR—The evaporator consisted of a single nickel tube, *C*, $\frac{3}{4}$ inch i. d., $\frac{7}{8}$ inch o. d., and 12 feet long. It was surrounded by a jacket, *B*, of 3-inch extra heavy iron pipe. The nickel tube was originally rolled into steel tube sheets at either end of this jacket, but it was found that the nickel was too soft to give a satisfactory grip, and short ferrules of monel metal were placed inside the tube at the tube sheets to give additional strength to the joint. A $1\frac{1}{2}$ -inch Kingsford pump, *P*, was located so that it discharged first through an orifice plate, *O*, and then into the bottom of the evaporator tube.

The top of the evaporator tube was connected to a vapor head, *A*, originally made of 14-inch flanged iron pipe, but later of cast iron. The inlet for the vapor into the vapor head was tangential, and from the bottom of the vapor head the liquid was withdrawn to the circulating pump. From the top of the vapor head the vapors passed through a surface condenser, *E*, connected to a vacuum pump. It was thought at one time that it would be desirable to operate the apparatus under a vacuum, but all the runs reported in this paper were made with atmospheric pressure in the evaporator. The condensate from the surface condenser was collected in calibrated drip receivers, *F*. The feed solution was contained in a sheet-monel storage tank, *Q*.

The diphenyl vapor was introduced into the top of the jacket around the evaporating tube by pipe *D*, and the condensate was withdrawn from the bottom of the jacket by pipe *U* and collected in drip receivers *N*. The drip tanks were

welded steel vessels, 14 inches in diameter and 30 inches high, provided with Jerguson high-pressure gage glasses. They were calibrated with water, and the calibrations converted to pounds diphenyl per millimeters by means of known densities of liquid diphenyl. From the bottoms of these drip receivers the condensed diphenyl could be returned to the boiler circulating pump and at the top were vent connections, *R* and *S*, leading to the flash chamber and to the melting tank. By equalizing these tanks with the flash chamber it was possible to pump them out with the diphenyl circulating pump.

The arrangement of the apparatus is shown in Figures 2 to 5.

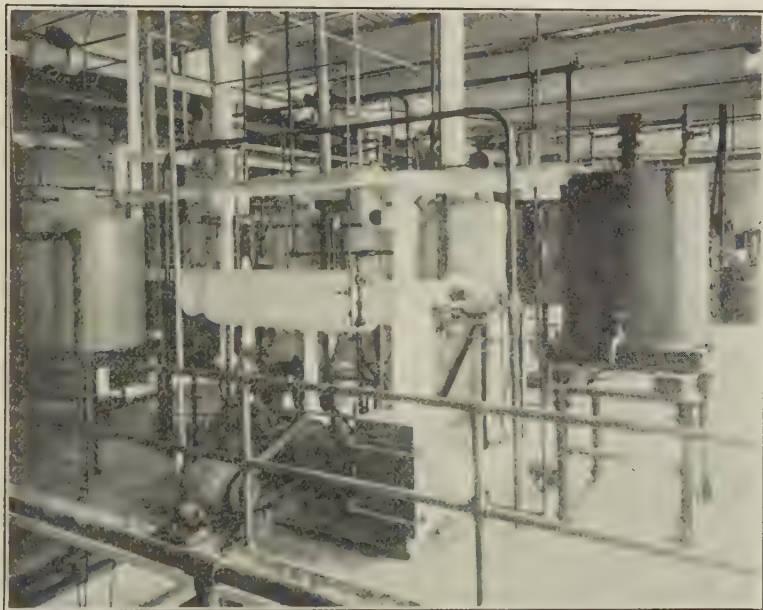


Figure 2—Rear View of Diphenyl Boiler

Figure 2 shows the heating tubes of the diphenyl boiler across the middle of the picture. The diphenyl circulating pump is inside the steel casing in the lower center; the diphenyl melting tank is at the extreme right, and the caustic storage tank at the extreme left. Figure 3 shows the operating space. The vessels from left to right are the diphenyl melting tank, the diphenyl flash tank, two diphenyl drip receivers, and beyond them the evaporating tube. The caustic circulating pump is shown in the right foreground.

Figure 4 is a view of the caustic circulating pump and the lower part of the evaporating tube. Figure 5 shows the upper part of the apparatus with the diphenyl vapor line at the extreme right, the top of the evaporator tube in the

center of the picture with the surface condenser behind it, and the vapor head of the evaporator to the left of the evaporator tube.

All parts in the diphenyl cycle were constructed of ordinary cast iron or steel. As stated above, the evaporator tube was nickel. The liquor circulating pump was provided with a nickel-cast-iron impeller and casing and a steel shaft. At the beginning of the work the liquor circulation lines were ordinary iron pipe. Cast-iron Merco cocks were used as valves. The vapor chamber was made from iron pipe with cast-iron flanges. After a certain period of operation, it was found that the caustic was badly contaminated with iron and that there were many leaks in the caustic circulating system.

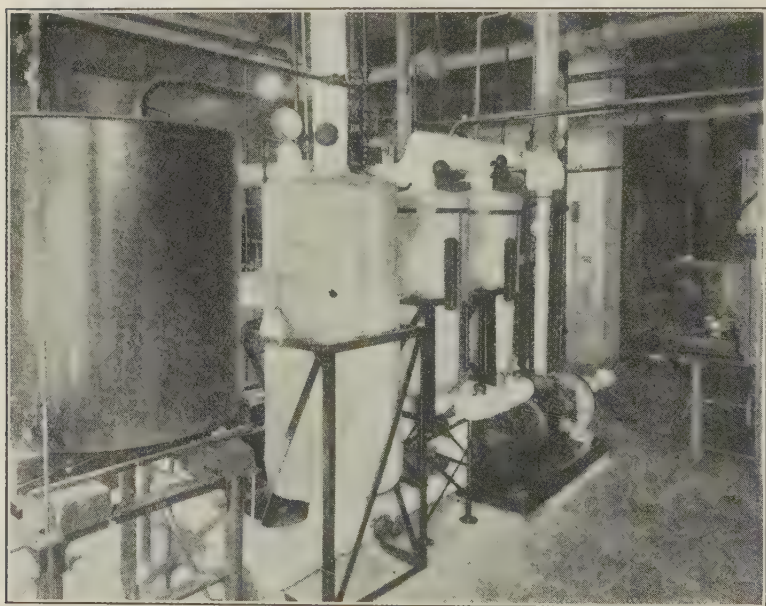


Figure 3—Diphenyl Boiler and Evaporator

It was found that the iron pipe was badly attacked and nearly eaten through in places. The vapor head was replaced with a nickel-cast-iron vapor head, and the iron piping was replaced with monel pipe. Elbows and reducers were still cast iron and the cast-iron Merco cocks were also retained. The final construction has satisfactorily withstood about a year's use, but has resulted in the caustic becoming somewhat contaminated with iron.

TEMPERATURE MEASUREMENTS—It was necessary to take measurements, not only of the temperature of the diphenyl vapor and the boiling caustic, but also the temperature of the tube wall. All these measurements were made by copper-

nickel thermocouples. A number of copper wires were plated to the evaporator tube by the method of Othmer and Coats (3), and one nickel wire was plated to the tube to serve as a common lead for all the couples. By connecting nickel leads to these copper wires junctions were obtained in the vapor space. One junction was placed in the liquor line just beyond the discharge from the tube and for a short time one was placed in the diphenyl drip line. These connections are shown diagrammatically in Figure 6.

A copper-nickel thermocouple was calibrated in the laboratory with samples of lead, tin, and zinc obtained from the Bureau of Standards and also with other standard melting points and boiling points. The curve so obtained is shown in Figure 7. This couple has been rechecked at various times and found to be constant within $\pm 1^\circ \text{F}$. This couple was inserted into the vapor space and compared with the other couples by turning high-pressure steam both into the jacket around the tube and into the inside of the tube itself. The steam space on both sides was vented sufficiently to insure the removal of non-condensable gases, and the steam pressure determined with an accurately calibrated gage. It was found that all the couples, including the calibrated one, gave results identical within $\pm 1^\circ \text{F}$. at all temperatures up to about 360°F ., and these temperatures checked the temperatures of the steam to within the same limit. Hotter diphenyl vapor at different pressures was used and the couples found to be the same up to 700°F . It was not possible to calibrate the other couples in place, but since they checked the calibrated couple within 1°F . at all temperatures, and since this calibrated couple was rechecked occasionally in the laboratory and found to be constant, it is believed that all the readings reported in this paper are accurate to the same limit—namely $\pm 1^\circ \text{F}$.

MECHANICAL DIFFICULTIES—In the caustic side of the system the difficulties due to corrosion have already been mentioned. The use of nickel and nickel-cast iron has apparently been satisfactory for the purposes of the laboratory up to the highest concentrations reached—namely, 98 per cent NaOH. The nickel parts appear to be unattacked. The nickel-cast-iron vapor head, the cast-iron fittings, and the steel shaft of the circulating pump were attacked. This attack was sufficient to discolor the caustic badly, but not sufficient to indicate an unduly short life of the parts. The nickel-cast-iron casing and impeller on the circulating pump were apparently unattacked after about a year's use. The cast-iron Merco cocks were a source of some difficulty, as they could be operated when hot but were practically impossible to manipulate when cold.

The only other source of trouble on the caustic side was the packing in the circulating pump. The only packing which would withstand the action of hot concentrated caustic was a laminated copper packing, John Crane No. 500. When two rings of this were used, backed with two or three rings of

asbestos packing, no difficulty was experienced in keeping the stuffing box tight over long periods of time.

On the diphenyl side the mechanical difficulties were not so great as had been expected. It was known that diphenyl was extremely fluid when hot and would penetrate through leaks that would not be shown by water or steam tests. The welds on the diphenyl chamber and the drip receivers leaked slightly at the start, but the leaks were easily peened shut with a hammer. The various joints in the pipe handling liquid diphenyl were first made with forged steel fittings, because it was thought that it would be necessary to weld all these

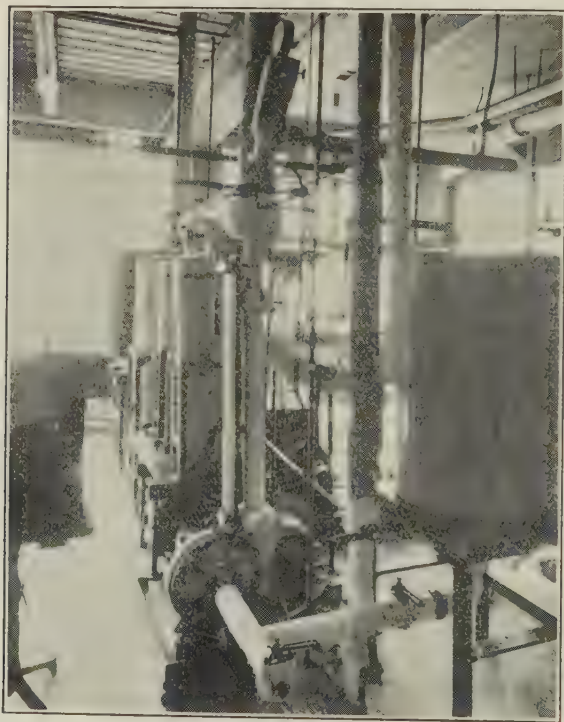


Figure 4—Caustic Evaporator

joints. As a matter of fact, it was not found difficult to make an ordinary screwed joint without welding that would hold diphenyl. This was accomplished by coating the threads with a litharge-copper oxide-glycerol cement. The valves in the diphenyl system gave considerable trouble. Ordinary 300-pound brass valves were used. In the sizes employed, the stuffing boxes were not satisfactory and leaked diphenyl vapor. A commercial installation should have more expensive valves with more carefully designed stuffing boxes, but other than leakage around the stem no difficulty was experienced.

Table I—Caustic Run

RUN	FINAL NaOH	AVERAGE DIPHENYL TEMP.	AVERAGE TUBE TEMP.	AVERAGE θ_v	LATENT HEAT DIPHENYL	DENSITY DIPHENYL	AVERAGE HEAT TRANSFER
	<i>Per cent</i>	$^{\circ}$ F.	$^{\circ}$ F.	$^{\circ}$ F.	<i>B. t. u. per lb.</i>	<i>Lbs. per cu. ft.</i>	<i>1000 B. t. per hour</i>
1	53.0	498	365	133	137	52.4	93.0
3	50.0	510	360	150	133	52.0	79.0
4	50.3	497	364	133	137	52.4	116.9
5	56.5	496	370	126	137	52.4	99.2
6	63.0	496	357	139	137	52.4	82.6
7	43.0	509	368	141	133	52.0	137.2
11	57.3	503	375	128	135	52.2	114.5
12	60.3	513	382	131	132	51.9	117.2
13	44.0	492	359	133	138	52.6	109.6
14	47.0	497	375	122	137	52.4	119.0
15	52.5	502	376	126	136	52.3	106.0
16	57.3	505	375	130	135	52.2	103.2
17	59.0	496	379	117	138	52.4	115.2
20	51.8	496	364	132	138	52.4	120.8
21	53.7	496	364	132	138	52.4	118.5
22	55.5	499	370	129	137	52.3	107.5
23	41.5	514	382	132	132	51.9	162.5
24	38.0	505	385	120	135	52.2	188.0
25	41.5	500	378	122	136	52.3	173.0
26	57.5	527	392	135	128	51.5	127.4
27	60.5	513	396	117	132	51.9	117.9
28	64.0	519	404	115	131	51.7	129.5
29	68.5	507	368	139	134	52.1	162.1
30	48.5	500	369	131	136	52.3	161.4
31	50.5	496	372	124	138	52.4	159.0
32	50.0	517	370	147	132	51.8	140.0
33	53.0	512	372	140	132	51.9	154.0
34	55.0	516	369	147	131	51.2	158.0
35	57.0	522	372	150	130	51.7	152.6
36	57.8	510	367	143	133	52.0	147.4
37	59.8	514	373	141	132	51.9	149.0
38	55.7	516	377	139	131	51.8	147.0
39	59.0	504	370	134	135	52.2	136.0
40	61.0	502	369	133	136	52.3	121.5
41	63.2	508	372	136	133	52.1	133.2
42	67.5	511	391	120	132	51.9	127.5
43	69.0	524	394	130	129	51.6	136.0
44	74.5	522	415	107	130	51.6	108.6
45	76.2	519	422	97	131	51.7	90.5
46	78.5	528	435	93	128	51.4	84.1
47	78.7	517	435	82	132	51.8	73.4
48	79.5	511	433	78	132	52.0	65.6
49	46.0	504	437	67	135	52.2	53.0
50	48.8	502	378	124	136	52.3	138.8

Evaporator—Series HG

FINAL PHENYL TEMP.	FINAL TUBE TEMP.	FINAL LIQUOR TEMP.	FINAL θ_v	FINAL HEAT TRANSFERRED	FINAL θ_{ni}	FINAL θ_L	FINAL U_L	VELOCITY
$^{\circ} F.$	$^{\circ} F.$	$^{\circ} F.$	$^{\circ} F.$	1000 B. t. u. per hour	$^{\circ} F.$	$^{\circ} F.$		<i>Fl. per sec.</i>
495	357	299	138	96.6	6	52	790	8.3
516	344	290	172	90.5	5	49	789	7.9
494	361	291	131	115.0	6	64	759	8.2
494	368	308	126	99.2	6	54	782	8.7
492	352	333	140	83.5	5	14	...	8.7
509	368	273	141	137.2	8	87	671	8.2
496	374	310	122	109.0	6	58	795	7.7
512	380	319	132	118.4	7	54	931	7.7
489	359	274	130	106.5	6	79	572	7.7
496	361	283	135	131.5	8	70	797	8.1
501	369	297	132	111.0	7	64	739	8.2
503	364	311	139	110.3	7	47	995	8.4
496	375	316	121	119.5	7	52	975	8.7
496	365	294	131	120.0	7	64	795	9.1
...	...	300	...	...	...	...	...	9.4
...	...	304	...	...	...	...	...	8.9
503	388	267	115	161.3	8	113	532	8.2
502	378	258	124	194.0	11	109	761	8.3
495	383	267	108	153.0	9	107	607	8.5
526	391	312	135	127.4	8	71	761	8.2
508	390	322	118	119.0	7	61	828	8.2
519	403	333	116	130.5	8	62	870	8.2
517	368	285	149	174.0	10	73	1010	...
503	375	285	128	158.9	9	81	826	8.2
495	369	290	126	161.9	10	68	993	8.2
513	370	290	143	136.1	8	72	803	8.0
513	365	298	148	163.0	10	57	1220	8.2
509	365	304	144	154.7	9	52	1260	8.2
509	368	310	141	143.5	8	50	1220	8.2
503	367	312	136	140.0	8	47	1268	7.9
516	375	317	141	149.0	9	49	1285	7.8
513	374	306	139	147.0	9	59	1055	6.5
504	369	316	135	137.0	8	45	1280	6.6
493	370	322	123	112.5	7	41	1165	5.3
503	385	370	118	115.9	7	48	1052	5.5
510	394	345	116	123.5	7	42	1245	7.3
516	396	352	120	125.0	7	37	1440	7.4
518	419	380	99	100.0	6	33	1292	7.4
514	425	391	89	83.0	5	29	1215	7.4
522	439	407	83	75.1	5	27	1180	7.4
505	434	407	71	63.5	4	23	1172	7.4
505	434	408	71	59.6	4	22	1155	7.3
498	439	413	59	46.6	3	23	860	7.3
508	389	279	119	133.0	8	102	552	7.6

The packing on the diphenyl circulating pump was the source of the greatest difficulty. The original design of this pump had a split stuffing box which was exceedingly difficult to keep tight. It is practically impossible to hold liquid diphenyl at high pressures with ordinary asbestos packing, but a small amount of cooling water freezes the diphenyl in the packing and makes a satisfactory seal. Later the split stuffing box was replaced with a single cast-iron stuffing box, and although this was tight to water at low temperatures it proved to have tiny leaks which became apparent at high temperatures and allowed water to leak into the diphenyl.

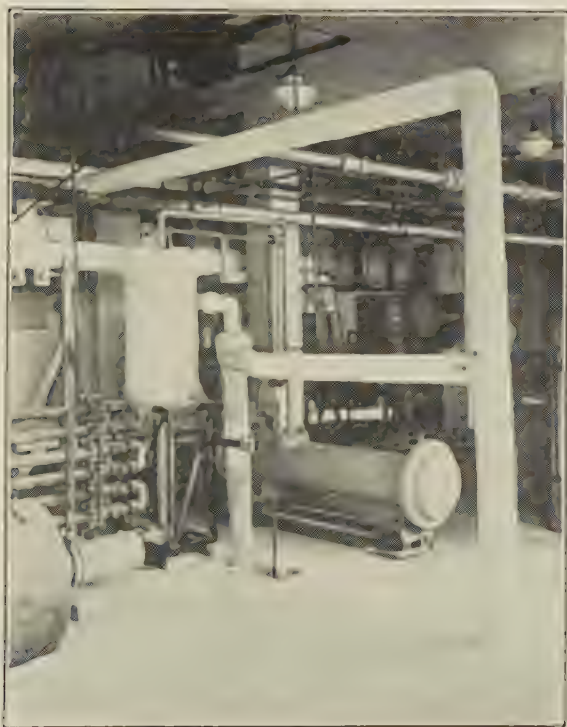


Figure 5—Top of Diphenyl-Heated Caustic Evaporator

A satisfactory seal was obtained only when a cast-bronze water-cooled stuffing box was provided. With this there has been no difficulty whatever in keeping the pump tight to diphenyl.

The diphenyl pump, as originally designed, gave continual difficulty at the joint in the casing because the studs and bolts were of poor quality and not on close enough centers. The same difficulty was experienced with the studs which held the Jerguson gage glasses to the diphenyl drip receivers. At the temperatures of operation these studs were too weak to stand the strain necessary to make tight joints and were easily

twisted off. Substituting studs and bolts turned from moderately high carbon steel solved this difficulty on both the gage glasses and the pump casing.

Considerable trouble was experienced from burning out of the electric heaters, but this was always due to a failure of the circulating pump, either from poor venting or from frozen diphenyl in the lines. Inasmuch as the electrically heated boiler is purely a laboratory makeshift and does not represent commercial design, this feature is not considered important.

All parts of the apparatus handling diphenyl are traced with high-pressure steam lines and heavily insulated. With

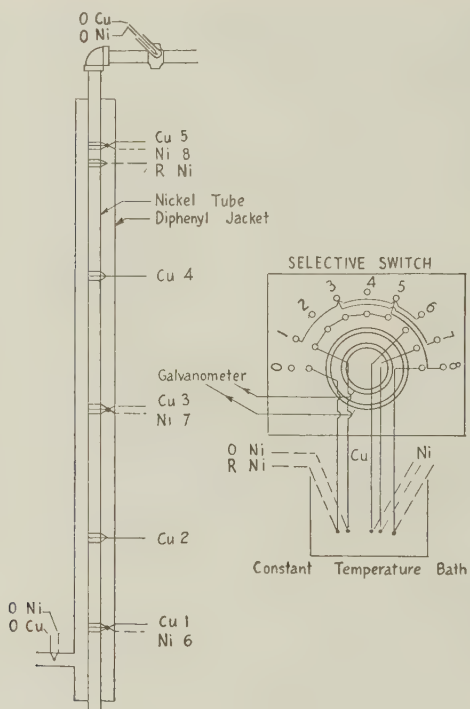


Figure 6—Junctions for Temperature Measurements

this provision it is not difficult to start the apparatus and to keep it free from freeze-ups.

RADIATION CORRECTIONS—Since the apparatus had a rather large external surface and a small heating surface, and since it was to operate at relatively high temperatures, the radiation correction became of importance. Radiation was determined by filling the vapor space with diphenyl vapor at a constant pressure and maintaining this until a drip receiver full of condensate had been obtained. Readings were taken at frequent intervals during this time. The radiation correction was expressed in millimeters of diphenyl

collected per minute in the drip receivers, and is plotted in Figure 8. It was found possible to obtain very satisfactory checks between duplicate determinations. Radiation runs were also made with both increasing and decreasing diphenyl temperatures in order to obtain the effect of changes in the heat capacity of the apparatus due to fluctuations in temperature during a run. It was found that unless the temperatures varied by more than $\pm 10^\circ \text{F}$. this effect was negligible.

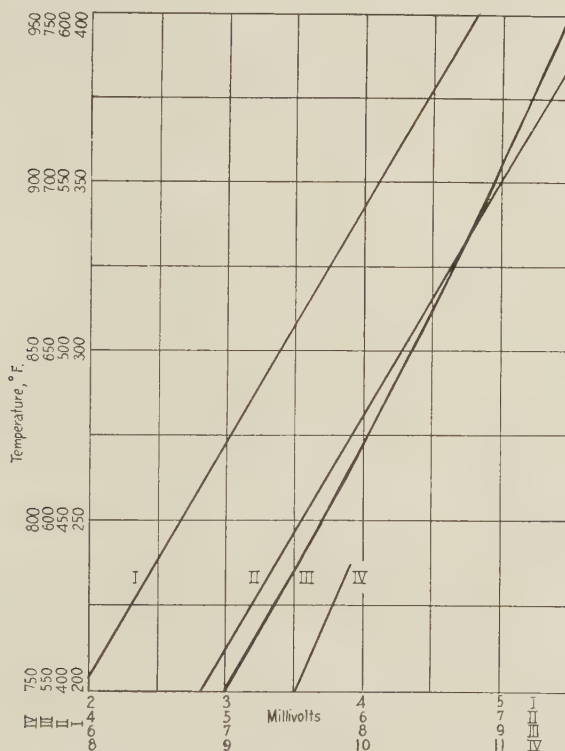


Figure 7

Method of Operation

Commercial flake caustic was dissolved to an approximately 50 per cent solution in the storage tank. This was about as strong as could be conveniently handled at room temperature without danger of freezing. The length of the individual experiments was usually kept short so that there would be no excessive change in concentration between the beginning and the end of the experiment. For a time the experiments were carried out during the day, but the delay in thawing out the caustic lines and the diphenyl lines and getting the apparatus up to temperature was so great that it became necessary to keep the apparatus hot through-

out the 24 hours, although the numerical data were largely taken during the day.

The calculation of the results presented certain difficulties. Total heat transferred was determined from the volume of condensed diphenyl collected in the drip tanks. No determinations were made of the quality of the diphenyl vapor as it reached the jacket of the evaporator. If it was not dry and saturated at this point, the coefficients as calculated are in error. The concentration of the caustic varied somewhat during a run and consequently its boiling point varied. Further, the actual temperature of the liquid over the length of the tube was higher than the boiling point, owing to the combined effects of hydrostatic head and friction. The difficulty of determining the mean liquid temperature was so great that it was considered out of the question for this

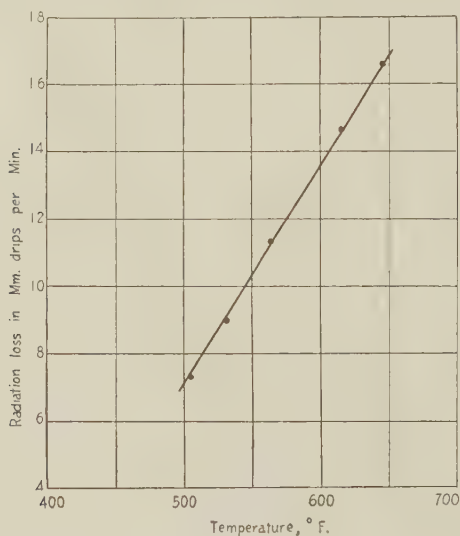


Figure 8

work, consequently the results were calculated on the basis of the boiling temperature of the caustic as it left the apparatus. The results, therefore, are apparent heat-transfer coefficients corrected for the elevation in boiling point of the solution. It was finally decided to base the calculations on the boiling temperature of the caustic at the end of a run rather than the average over the whole run, largely for the sake of simplifying the calculations. The change in concentration during the short runs employed was small and therefore the error so introduced was not great. The coefficients thus determined are more useful in design than the more scientific true coefficients. These results cover such a wide range of concentrations that the data are sufficient for any ordinary purpose of design.

In the first experiments no thermocouple was placed in the

Table II—Caustic

RUN	FINAL NaOH	AVERAGE DIPHENYL TEMP.	AVERAGE TUBE TEMP.	AVERAGE θ_v	LATENT HEAT DIPHENYL	DENSITY DIPHENYL	AVERAGE HEAT TRANSFER
	<i>Per cent</i>	<i>° F.</i>	<i>° F.</i>	<i>° F.</i>	<i>B. t. u. per lb.</i>	<i>Lbs. per cu. ft.</i>	<i>1000 B. per h.</i>
1	51.7	492	363	129	138	52.6	104
2	55.5	498	368	130	137	52.4	121
3	80.2	526	449	77	128	51.5	50
4	77.3	523	433	90	129	51.6	53
5	78.5	521	431	90	130	51.7	69
6	79.9	536	455	81	125	51.2	63
7	82.6	517	456	61	131	51.8	39
8	84.3	519	474	45	130	51.7	29
10	80.3	509	446	63	133	52.2	69
11	81.2	535	463	72	125	51.3	70
12	82.2	503	454	49	135	52.2	39
13	83.3	510	464	46	133	52.0	29
14	83.0	513	468	45	132	52.0	25
15	82.8	499	460	39	136	52.3	33
16	83.0	502	459	43	135	52.2	28
17	82.6	500	455	45	136	52.3	24
18	84.6	512	477	35	121	50.8	16
19	84.7	499	472	27	136	52.4	17
20	84.8	496	472	24	137	52.5	8
21	85.1	501	478	23	136	52.3	6
22	86.1	539	505	34	124	51.1	18
23	87.0	543	514	29	123	51.1	19
24	87.7	544	520	24	122	50.9	18
25	87.7	549	526	23	121	50.8	12
26	88.4	554	531	23	119	50.6	17
27	89.2	561	530	24	117	50.4	15
29	61.13	500	385	115	136	52.3	103
30	79.5	497	389	108	137	52.4	89
31	66.4	510	403	107	132.5	52.0	71
32	83.3	536	489	47	126.0	51.3	47
33	82.7	499	462	37	136	52.3	33
38	82.2	531	467	64	127	51.4	59
39	82.8	542	483	59	123	51.0	65
42	79.0	500	446	54	136	52.3	41
43	79.5	495	446	49	138	52.5	32
44	82.5	500	461	39	136	52.3	28
45	84.0	527	480	47	129	51.4	31
48	85.0	556	496	60	119	50.5	60
53	88.7	550	531	29	121	50.7	18
54	89.2	516	543	33	114	49.9	21
56	90.3	597	564	33	110	49.2	18
57	91.0	587	561	26	112	49.5	12
58	91.5	606	570	36	108	48.8	15
59	60.0	549			121	50.7	147
60	64.0	554			119	50.6	146
61	70.0	573			114	50.0	134
62	76.5	566			116	50.2	106
63	80.0	566			116	50.2	109
64	82.6	572			115	50.0	120
65	85.0	570			115	50.1	83

orator—Series JU

FINAL TUBE TEMP.	FINAL LIQUOR TEMP.	FINAL θ_v	FINAL HEAT TRANSFERRED	FINAL θ_{ni}	FINAL θ_L	FINAL U_L	VELOCITY
$^{\circ} F.$	$^{\circ} F.$	$^{\circ} F.$	1000 B. t. u. per hour	$^{\circ} F.$	$^{\circ} F.$		<i>Ft. per sec.</i>
390	294	122	98.0	6	70	639	7.2
376	306	126	117.5	7	63	848	7.4
487	420	69	57.6	3	34	705	8.9
431	397	88	52.5	3	32	755	7.9
430	403	80	61.5	4	27	1037	5.7
458	417	77	60.4	4	37	748	6.6
465	446	51	33.0	2	17	884	6.9
479	461	44	29.0	2	16	825	6.2
451	420	59	64.7	4	27	1086	6.0
461	429	63	61.6	4	31	908	5.6
463	439	42	33.5	2	22	694	5.2
470	450	39	24.7	2	18	624	5.1
473	456	43	24.4	2	21	530	5.8
468	448	34	29.2	2	18	737	5.6
466	448	33	21.9	1	17	586	4.7
463	446	40	21.3	1	15	652	4.0
481	466	32	15.2	1	14	493	4.7
477	470	21	13.5	1		1025	3.9
478	470	19	6.6	1	7	425	3.2
479	472	20	5.9	1	6	453	2.2
508	486	31	17.1	1	21	375	2.7
516	500	27	16.7	1	15	426	4.0
524	509	23	17.6	1	14	538	3.9
526	516	18	13.5	1	9	683	2.3
534	520	21	15.6	1	13	549	2.6
540	531	20	13.0	1	8	738	2.5
	Av.			Av.	Av.	Av.	
	318			6	54	807	
	333			5	51	742	8.5
	342			4	57	530	7.9
	452			3	34	596	13.6
	445			2	15	954	9.4
	439			4	24	1060	5.5
	446			4	33	835	6.2
	410			3	33	577	7.2
	413			2	31	485	7.4
	444			2	15	827	
	460			2	18	789	
	470			4	20	1375	
	524			1	6	1400	
	531			1	11.5	845	
	548			1	15	565	
	551			1	9	625	
	557			1	12	585	
	316						
	332						
	355						
	406						
	418						
	441						
	469						

caustic as the primary interest was in the diphenyl film coefficients. At the end of a run the supply of diphenyl vapor was shut off and the tube allowed to come to liquid temperature. In later runs a thermocouple was placed in the liquid just beyond the evaporator tube, and temperatures were taken during the run. These two methods checked very well. Especially at the higher concentrations, it was not feasible to remove samples for specific gravity determinations, and the number of runs was so great that it was not considered feasible to determine the concentration by analysis. The boiling points of strong solutions have been reported by Gerlach (2), and by von Antropoff and Sommer (1). These data are given in Figure 9. The data of Gerlach were used up to 50 per cent and the data of von Antropoff and Sommer above 50

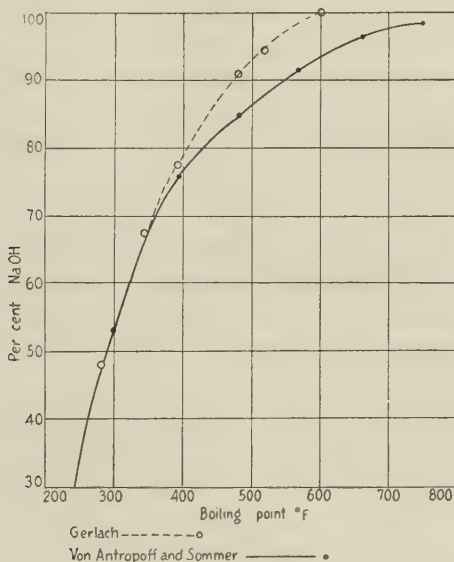


Figure 9

per cent. Concentrations were calculated from the observed boiling point and the above data. The operation of the evaporator and the data taken are obvious from Figure 1.

Heat-Transfer Coefficients

No attempt was made to calculate coefficients when the temperature differences were less than 10°F. , as the accuracy of the thermocouples is too low to give such results any precision. All runs in which the pressure of the diphenyl was less than atmospheric were discarded, as a slight leakage of air into the diphenyl space would seriously affect the results. The data for some of the runs are contained in the accompanying tables. The headings of the various columns are self-explanatory with the exception of certain mathematical symbols which have the following significance:

θ_v = temperature drop between diphenyl vapor and outside surface of tube
 U_v = diphenyl film coefficient in B. t. u. per square foot per hour per degree Fahrenheit
 θ_{ni} = temperature drop through metal of tube, in degrees Fahrenheit

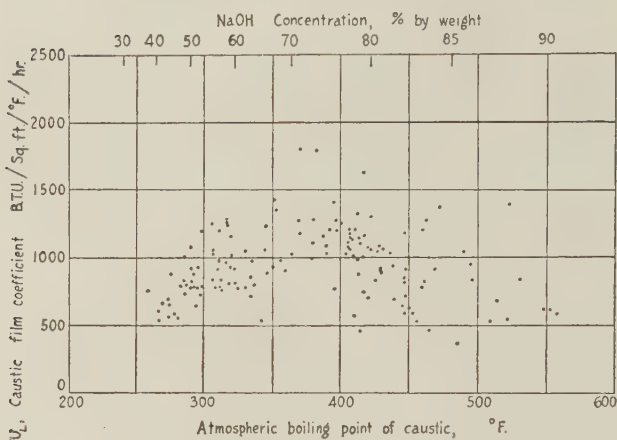


Figure 10

θ_L = temperature drop between inside surface of tube and circulating liquid
 U_L = caustic film coefficient in B. t. u. per square foot per degree Fahrenheit per hour

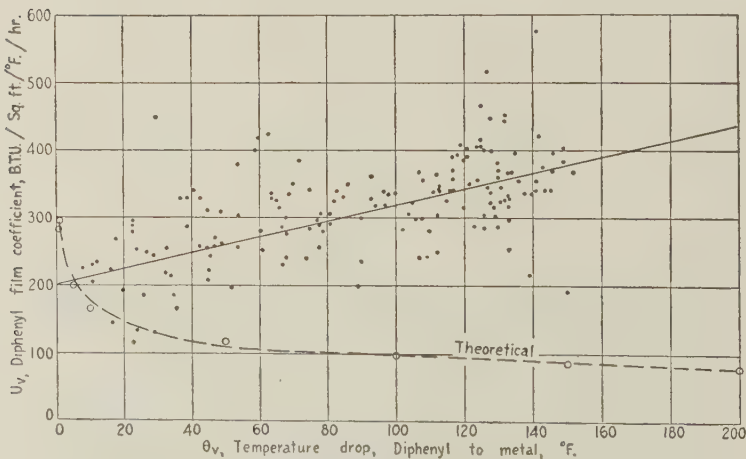


Figure 11

It is believed that the accuracy of the coefficients reported in Tables I and II and plotted in Figures 10 and 11 was approximately 5 per cent. The spattering of the points for the liquid film coefficient in Figure 10 is probably due to variations in the conditions of velocity, temperature drop, purity of solution, etc.

Figure 11 shows the diphenyl film coefficient plotted versus temperature drop. No marked effect of temperature of the vapor was noted, but the coefficient was found to increase with increasing temperature difference. The spattering of the points and the application of Nusselt's theory of vapor condensation (dotted line) is explained in the article on the condensation of vapors.²

An interesting feature of this work is the fact that, under the conditions employed, the liquid film coefficient is so high compared with the diphenyl film coefficient that the latter controls. In runs JU 59 to 65 some of the thermocouples were broken so that only over-all coefficients could be determined, and it is interesting to note that these are only slightly lower than the diphenyl film coefficients.

Concentration Obtained

In these experiments caustic was actually concentrated to 98.2 per cent. No data are reported for concentrations in excess of 92 per cent, however, because at the diphenyl pressures employed for the higher caustic concentrations (about 150 pounds gage) it was practically impossible to keep the gage glasses on the drip receivers tight, and consequently the diphenyl condensate was returned directly to the boiler and no measurements could be made.

Considering that the whole apparatus was evolved *de novo*, with no precedent or previous experience, and that in spite of this caustic was almost completely dehydrated, it is believed that a properly designed commercial apparatus for concentrating caustic soda with diphenyl would be entirely practical. Aside from the information on heat-transfer coefficients which are reported in this paper, the principal result is that caustic has been dehydrated in a continuous apparatus with entirely negligible mechanical difficulties.

Acknowledgment

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